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Liquid and flow boiling heat transfer inside a copper foam

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Abstract

This paper presents the experimental measurements carried out during liquid and flow boiling heat transfer of R1234ze(E) and R134a inside a high porosity copper foam. The test section with a 5 pores per linear inch (PPI) foam, is electrically heated from the bottom and it is instrumented with 20 thermocouples to monitor the wall temperature distribution at different imposed heat fluxes, saturation temperatures, vapour qualities, and refrigerant mass flow rates. The experimental measurements were conducted in a new experimental facility built at the Dipartimento di Ingegneria Industriale of the Università di Padova especially designed to study either liquid or two-phase heat transfer processes for electronic thermal management applications.

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Keywords: Flow boiling; R1234ze(E), R134a, Heat Transfer, Pressure drop, Two-Phase flow

1. Introduction

The possible use of a low-GWP refrigerant as working fluid would be an important feature for an efficient, eco-friendly, and smart cooling system for electronic thermal management. Over the last several years, much research and development effort has been focused on potential refrigerants possessing low GWPs. Among the fluorinated propene isomers which have normal boiling point temperature data published in the public domain, several have low GWP and normal boiling temperatures relatively close to R134a; among them, R1234ze(E) has a normal boiling temperature approximately 7.3 °C lower than that of R134a. It has a GWP<1 and is being widely considered as a possible replacement for R134a in different applications, including electronic thermal management, air conditioning and refrigeration systems.

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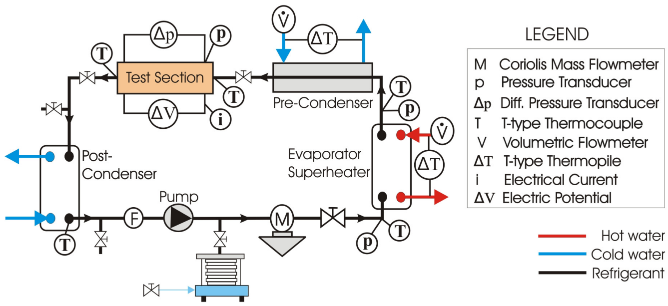


Figure 1. Schematic of the experimental setup.

Boiling is the heat transfer mechanism with the highest heat transfer coefficients, thus it can be used to spread high fluxes to maintain the junction temperature at low values with compact heat sinks. Moreover, new surfaces with microporous coatings, Carbon Nano Tubes (CNTs) coatings, or microstuctured surfaces are now available to enhance the boiling phenomenon. Nevertheless, a lot of work is still needed to deeply understand the boiling mechanism in such surfaces, where a huge number of variables (among those heat flux, saturation temperature, flow pattern, gravity, subcooling, wall surface, and others) are linked together and play important and crucial roles.

By virtue of their interesting multifunctional properties, open-cells metal foams represent promising alternative enhanced surfaces also for flow boiling applications. Nowadays, almost all the research is focused on single-phase flow in metal foams (air or liquid flow) and only few works regard the phase change process. Mancin et al. (2013) carried out a comprehensive work on single-phase heat transfer through metal foams, while a comprehensive review of metal foam heat transfer was proposed by Zhao (2012). Among the few works on flow boiling heat transfer in metal foams the most interesting appear to be: Zhao et al. (2009), Kim et al. (2008), Li and Leong (2001), and Ji and Xu (2012).

However, no one investigated the flow boiling heat transfer of the new low-GWP refrigerant R1234ze(E). This work aims at investigating the heat transfer properties of R1234ze(E) fluid during both liquid and flow boiling inside a copper foam.

2. Experimental Setup and Data Reduction

The experimental set up is located at the Heat Transfer in Micro-Geometries Lab (HTMg-Lab) of the Dipartimento di Ingegneria Industriale of the Università di Padova. As shown in Fig. 1, the experimental facility consists of three loops: the refrigerant, the cooling water, and the hot water loop. The rig was designed for heat transfer and pressure drop measurements and flow visualization during either vaporization or condensation of pure refrigerants and refrigerants mixtures inside structured micro-geometries. The facility has a maximum working pressure of 3 MPa, while refrigerant mass fluxes can be varied up to 400 kg m^{-2} s^{-1} in a section of 50 mm^2. In the first loop the refrigerant is pumped through the circuit by means of a magnetically coupled gear pump, it is vaporized and superheated in a brazed plate heat exchanger fed with the hot water.

Superheated vapour then partially condenses in a precondenser fed with the cold water to achieve the set vapour quality at the inlet of the test section. The refrigerant enters the test section at a known mass velocity and vapour quality and then it is vaporized by means of a calibrated Ni-Cr wire resistance.

The fluid leaves the test section and passes through a post-condenser, a brazed plate heat exchanger, where it is fully condensed and subcooled. Then, the subcooled liquid is sent back to the boiler by a pump.

A damper connected to the compressed air line operates as pressure regulator to control the saturation condition in the refrigerant loop. As shown in Fig. 1, the refrigerant pressures and temperatures are measured in several locations throughout the circuit to know the refrigerant properties at the inlet and outlet of each heat exchanger.

The refrigerant flow can be independently controlled by the gear pump and it is measured by means of a Coriolis effect flowmeter. The inlet vapour quality to the test section is determined by the heat extracted in the precondenser, which can be controlled by varying water temperature and flow rate. The cold water loop consists of a stabilized chiller connected to the precondenser. The hot water circuit consists of a pump, an electrical heater and a controlling valve; it permits to set both the water flow rate and the inlet water temperature. Water flow rates in the precondenser and boiler sections are measured by means of magnetic type flow meters, while the water temperature differences are measured using 4-junction T-type thermopiles.

The test section consists of a 440x130x50 mm of low conductivity MISOGLASS block. This material has been selected by virtue of its interesting properties: high working temperature (greater than 200 °C), low thermal conductivity (around 0.35 W m^{-1} K^{-1}). The block is machined to obtain two plenums where the refrigerant enters and exits and where both the refrigerant temperature and pressure are measured by means of calibrated T-type thermocouples and high accuracy pressure transducers, respectively. A guide is milled in the center of the block to locate the heater and the foam sample.

The foam sample is 10 mm wide, 5 mm high, and 190 mm long, it has 5 PPI and a porosity ε (defined as the ratio between the void volume and the solid and void total volume) of 0.93; it has been brazed over a 200x10x10 mm of copper plate. 20 holes (5 mm deep) have been drilled just 1 mm under the foam-wall brazing surface to monitor the wall temperature distribution by locating as many calibrated T-type thermocouples.

Considering the data reduction, in the case of flow boiling heat transfer measurements, the vapour quality at the inlet of the test section depends on the refrigerant conditions at the inlet of the precondenser and from the heat flow rate exchanged in the tube-in-tube heat exchanger which can be obtained from a thermal balance on the cooling water side as given by:

$$q_{pc} = \dot{m}_{w,pc} \cdot c_{p,w} \cdot \left(t_{w,pc,out} - t_{w,pc,in}\right) = \dot{m}_{ref}\left(h_{vs} - h_{TS,in}\right) \tag{1}$$

where h_{vs} is the enthalpy of the superheated gas at the inlet of the precondenser. The vapour quality at the inlet of the test section x_{in} can be calculated from the value of the enthalpy at the inlet of the test section $h_{TS,in}$ by applying a heat balance as described by Eq. (1), as:

$$x_{in} = \frac{h_{TS,in} - h_L}{h_V - h_L} \tag{2}$$

where h_L and h_V are the specific enthalpies of the saturated liquid and vapour, respectively, evaluated at saturation pressure of the refrigerant measured at the inlet of the test section.

The outlet vapour quality is obtained from a heat balance applied to the test section accounting for the heat losses. In particular, the electrical power P_{EL} supplied to the sample is indirectly measured by means of a calibrated reference resistance (shunt) and by the measurement of the effective EDP (Electrical Difference Potential) of the resistance wire inserted in the copper heater. The current can be calculated from the Ohm's law. Preliminary heat transfer measurements permitted to estimate the heat losses (q_{loss}) due to conduction through the test section as a function of the mean wall temperature; the heat lost through the test section was always less than 3%. The two-phase heat transfer coefficient HTC_{TP} is referred to the actual heat flow rate exchanged q_{TS}, as:

$$HTC_{TP} = \frac{q_{TS}}{A_{base} \cdot (\bar{t}_{wall} - \bar{t}_{sat})} = \frac{P_{EL} - q_{loss}}{A_{base} \cdot (\bar{t}_{wall} - \bar{t}_{sat})} \tag{3}$$

where $\bar{t}_{wall}$ is the average value of the measured wall temperatures and $\bar{t}_{sat}$ is the average value of the saturation temperatures obtained from the measured values of the pressure, as:

$$\bar{t}_{wall} = \frac{1}{20}\sum_{i=1}^{20} t_{wall,i} \qquad \bar{t}_{sat} = \frac{t_{sat,in}(p_{sat,in}) + t_{sat,out}(p_{sat,out})}{2} \tag{4}$$

Considering the liquid heat transfer measurements, in this case the heat transfer coefficient is defined as:

$$HTC_{SP} = \frac{q_{TS}}{A_{base} \cdot \dfrac{(t_{wall,in} - t_{ref,in}) - (t_{wall,out} - t_{ref,out})}{\ln \dfrac{(t_{wall,in} - t_{ref,in})}{(t_{wall,out} - t_{ref,out})}}} \quad (5)$$

where $t_{wall,in}$, $t_{wall,out}$ and $t_{ref,in}$, $t_{ref,out}$ are the wall and refrigerant temperatures measured at the inlet and outlet of the test section, respectively.

Thermodynamic and transport properties are estimated from RefProp v9.1 (Lemmon et al., 2013). The T-type calibrated thermocouples present an accuracy of ±0.05 K, while the electrical power supplied to the test section has an uncertainty of ±0.13%. From the error propagation analysis, it is estimated that the uncertainty on the two-phase heat transfer coefficient shows a mean value of ±2.5% and a maximum values of ±3.8%, while the uncertainty on the vapour quality is ±0.035. The single-phase heat transfer coefficient has an uncertainty of ±3%. The inlet absolute pressure and the pressure drop across the test section are measured with high accuracy absolute (±1950 Pa) and differential (±25 Pa) pressure transducers, respectively.

3. Experimental Results

The following figures show the R1234ze(E) flow boiling results carried out at 30 °C of saturation temperature. Figures 2 and 3 report the effects of the mass velocity and of the heat flux on the two-phase heat transfer coefficient as a function of the vapour quality, respectively. Considering Figure 2, the HTC_{TP} increases as the vapour quality increases and seems to be independent on the mass flux up to 75 kg m^{-2} s^{-1}, meaning that the nucleate boiling dominates the phase change process. Then, HTC_{TP} increases with the refrigerant mass velocity, meaning that the two-phase forced convection is becoming more and more the dominant phase change mechanism. The critical vapour quality at the onset of the dryout does not depend on the mass velocity and it is almost constant ranging between 0.76 and 0.78. After this point, the HTC_{TP} suddenly decreases and all the mass velocities show similar trends of the measured values.

Figure 3 shows the effects of the imposed heat flux on the flow boiling heat transfer at G=100 kg m^{-2} s^{-1}. At 50 and 75 kW m^{-2}, the HTC_{TP} are almost similar, while when increasing the heat flux up to 100 kW m^{-2}, the heat transfer coefficient also increases, especially at low vapour qualities; moreover, the vapour quality on the onset of the dryout passes from around 0.7 at 50 kW m^{-2} to 0.60 at 100 kW m^{-2}.

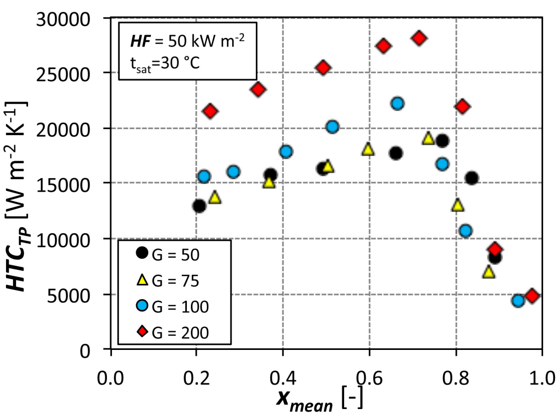


Figure 2. HTC_{TP} vs. vapour quality as a function of the mass flux. *G* expressed in [kg m^{-2} s^{-1}].

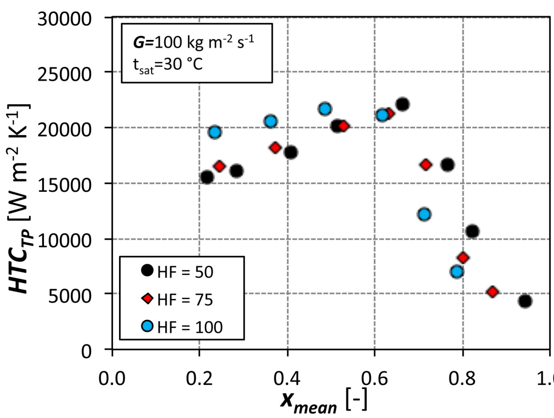


Figure 3. HTC_{TP} vs. vapour quality as a function of the heat flux. HF expressed in [kW m^{-2}].

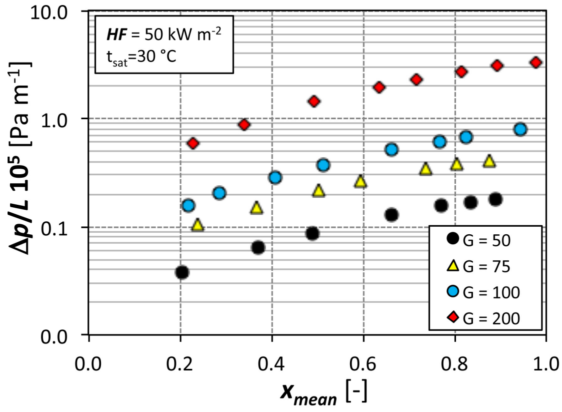


Figure 4. Two-phase pressure gradients vs. vapour quality as a function of the mass flux. *G* expressed in [kg $m^{-2}s^{-1}$].

Figure 4 reports the measured values of the pressure gradients as a function of the vapour quality at 50 kW m^{-2}. The pressure drop increases with both the vapour quality and the mass velocity. Finally, it is interesting to point out that different heat transfer and pressure drop measurements were also carried out at 40 °C of saturation temperature, but no noteworthy differences were found, for this reason and, for the sake of brevity, the results relative to this temperature are not here proposed.

Figures 5 and 6 compare the heat transfer and fluid flow behaviours of R1234ze(E) and R134a, at constant G=100 kg m^{-2} s^{-1} and HF=100 kW m^{-2}, at t_{sat}=30 °C. At these operating conditions, the two refrigerants show almost the same HTC_{TP} but the new low-GWP fluid exhibits higher two-phase pressure gradients; besides, the onset dryout occurs earlier for R1234ze(E) than for R134a.

Figures 7 and 8 compare the heat transfer coefficient and pressure gradient measured during single-phase liquid flow of R1234ze(E) and R134a inside the 5 PPI copper foam. The tests were run imposing a heat flux of 25 kW m^{-2} with the liquid entering the test section with at least 15 K of subcooling. As already found in the case of two-phase flow, the low-GWP refrigerant R1234ze(E) shows similar heat transfer coefficients but, at the same operating test conditions, its pressure drops are higher than those measured for the traditional R134a.

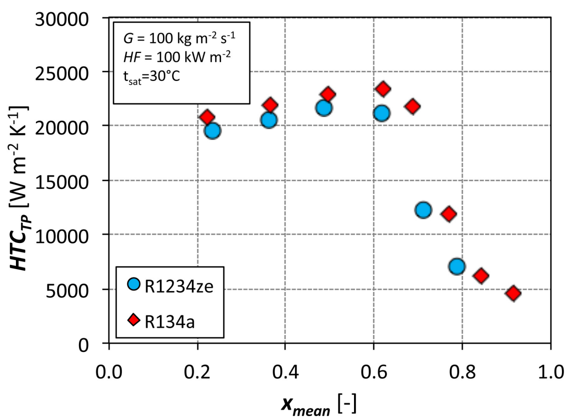


Figure 5. Comparison between R1234ze(E) and R134a, HTC_{TP} vs. mean vapour quality.

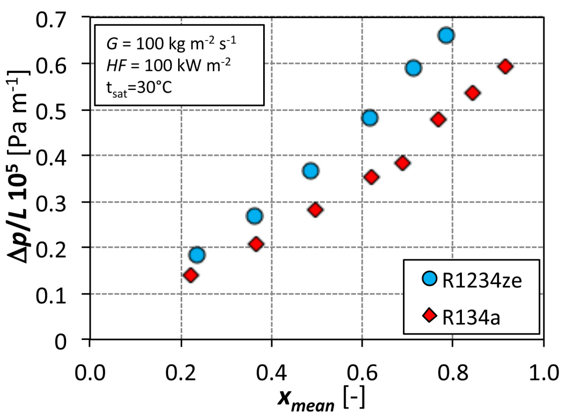
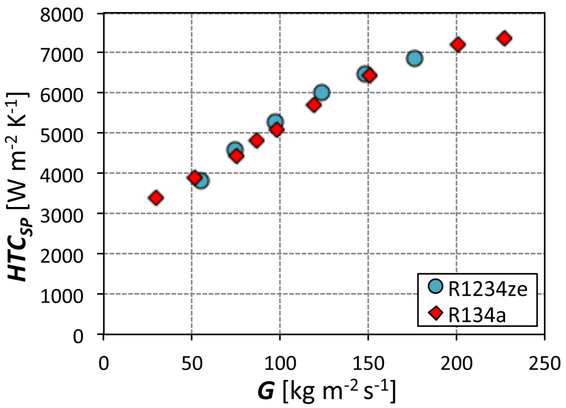


Figure 6. Comparison between R1234ze(E) and R134a, pressure gradient vs. mean vapour quality.



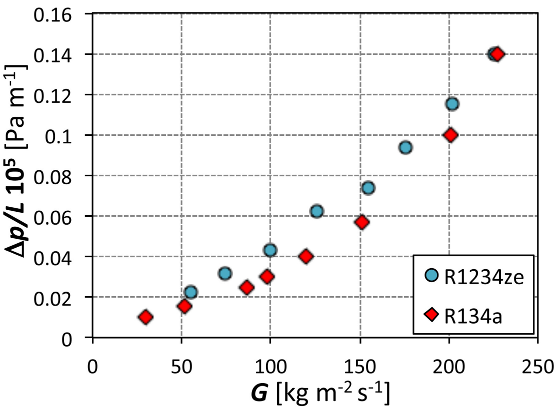


Figure 7. Comparison between R1234ze(E) and R134a, single-phase HTC_{SP} vs. mass flux.

Figure 8. Comparison between R1234ze(E) and R134a, single-phase pressure gradient vs. mass flux.

4. Conclusion

This paper presents some experimental results relative to flow boiling heat transfer and pressure drop of the new low-GWP R1234ze(E) inside an open cell metal foam with 5 PPI and porosity of 0.93. The measurements were carried out at constant saturation temperature of 30 °C, by varying heat flux, mass velocity and vapour quality. Moreover, the R1234ze(E) has also been tested during the single-phase liquid heat transfer through the porous medium. The single- and two-phase heat transfer and pressure drop results have been compared against those previously obtained for R134a. This work represents the first step towards the understanding of the real heat transfer capabilities of the new low-GWP R1234ze(E) refrigerant, which is a possible substitute of the R134a in advanced cooling systems. Finally, the results here presented demonstrate the interesting heat transfer capabilities of metal foams during the flow boiling and liquid heat transfer; more experimental work is needed to deeply understand the two phase heat transfer process inside these new enhanced surfaces.

5. Acknowledgement

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